

Dear Sir:

This responds to the Office Action mailed on July 29, 2003.

REMARKS

A. 35 U.S.C. § 102

1. Claims 12-32 and 37-53

Claims 12-32 and 37-53 were rejected under 35 U.S.C. §102(b) as being anticipated by Feichtinger et al. Applicant traverses this rejection. In particular, independent claims 12 and 37 each recite a coupling element that has two brackets wherein flexural strength of each bracket in a region along an axis of rotation between an inner support connection location of the respective bracket and a connecting line of first and second outer support connection locations of the respective bracket is considerably greater than the flexural strength in a region along a circumferential direction between the inner support connection location and the first and second outer support connection locations of the respective bracket. A detailed analysis of Feichtinger et al. is presented below that reveals that the flexural strength along the axis of rotation as recited in claims 12 and 37 is not present.

As shown in FIG. 2, the coupling 2 of Feichtinger has two spring parallelograms, at right angles to one another, in the form of leaf spring arms 6 and 7 and also 8 and 9. These leaf spring arms 6, 7; 8, 9 extend parallel to the pivot axis D, as shown in FIG. 5, and are diametrically opposite one another in pairs 6, 7; 8, 9 and parallel to one another. The resilience or deflectability in the radial direction (perpendicular to the pivot axis D) is based on the principle of the radial deflectability

of the leaf spring arms 6-9, because these arms are deflected radially over their axially parallel length.

It is well known in the art that Feichtinger et al.'s use of spring parallelograms inherently invokes several principles. First, the flexural strength of the leaf spring arms is as slight as possible between the central support point (fastening point at the base) and the connecting line of the two outer support points of a leaf spring arm. Second, the geometric dimensions of the leaf spring arms are chosen such that a deflection between the central support point and the connecting line of the further, outer support points can take place, and bending of the leaf spring arm between the two outer support points -- that is, in the circumferential direction, is avoided.

With the above kept in mind, an analysis of the Feichtinger et al. coupling 2 is now made where it is assumed for exemplary purposes that the distance between the central support point and the connecting line of the further, outer support points is about 9 mm. Assuming that the scale shown in the drawings is correct, then the distance of the two outer support points from one another is about 28 mm based on the above distance of about 9 mm as shown by the attached marked up versions of FIGS. 4 and 5 of Feichtinger et al. For a force F (represented by line F in FIG. 4 and a cross within a circle in FIG. 5 denoting that the force is perpendicular to the plane of the page) transmitted to the stator 10 and material conditions EJ , an area of the leaf spring arm with a length, l , of 9 mm is deflected in the same direction as force F by a spring deflection f_l (represented by lines f_l in attached Fig. 4) in accordance with the

following formula¹ for a spring fastened on one end since this leaf spring arm connects together the two components that can be deflected radially to one another:

$$f_1 = Fl^3/3EJ.$$

Note that the deflection f_1 occurs at an area spaced apart by 9mm from the point labeled “central support point” in attached FIG. 5.

As for the spring region between the two further support points, the formula for the spring travel, f_2 , given a force F and material conditions EJ , is the formula for springs firmly fastened on both ends:

$$f_2 = Fl^3/192EJ$$

From this it can be seen that for identical force conditions F and material conditions EJ , the spring travel f_1 ($= 9^3/3 * F/EJ = 243 * F/EJ$) for the free leaf spring arm having the length of 9 mm is more than twice as long as the spring travel f_2 ($= (27)^3/192 * F/EJ = 103 * F/EJ$) between the two further support points. In other words, the flexural strength of the leaf spring arm that is 9 mm long is less than half the flexural strength of the spring region between the two further support points, so that when a force F occurs, the bending of the leaf spring arm that is 9 mm long ensues, that is, the bending of the spring parallelograms. During such bending, the deflection value for f_1 is prevalent. When the force F acts, first the spring arm with

¹ See enclosed copy of a pertinent section of “Feinmechanische Balemente” by Prof. Dr. –Ing. Siegfried Hildebrand.

the least resistance, i.e., with the least flexural strength, yields, so that in actuality the deflection f_1 occurs almost exclusively since the deflection f_2 between the two outer support points labeled in attached FIG. 5 is negligible. Thus no deflection and no expansion between the two outer support points occur. This fact, documented above by the above calculations, is also logical, since Feichtinger et al. seeks radial deflection via parallel deflection of one spring parallelogram at a time, that is, two spring arms disposed parallel to one another.

From the description above, Feichtinger et al. does not disclose a bracket that has a flexural strength along an axis of rotation between an inner support connection location of the bracket and a connecting line of first and second outer support connection locations of the respective bracket that is considerably greater than the flexural strength in a region along a circumferential direction between the inner support connection location. Since Feichtinger et al. fails to disclose using the recited flexure, the rejection is improper and should be withdrawn.

Besides not being anticipated by Feichtinger et al., the claims are not rendered obvious by Feichtinger et al. since it does not suggest using a bracket with the flexure properties recited in independent claims 1 and 37. Without such motivation, the claims should be allowed.

2. Claims 33-36 and 54-57

Claims 33-36 and 54-57 were rejected under 35 U.S.C. §102(b) as being

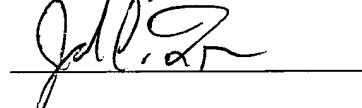
anticipated by Feichtinger et al. Applicant traverses this rejection. In particular, independent claims 33 and 54 each recite a coupling element that has two pairs of brackets wherein each pair of brackets includes outer support locations and inner support connection locations are located on a common plane that extends at a right angle with respect to the axis of rotation. The Office Action has relied on three locations labeled as 19a, 19b and C in a marked up version of FIG. 2 as identifying the outer and inner support locations of a spring arm 6. The Office Action has also identified spring arms 8a and 8b as being the outer and inner support locations of the spring arm 8. However, locations 19a, 19b, C, 8a and 8b are not located in a common plane that is at a right angle with respect to an axis of rotation as required by claims 33 and 54 and so the rejection is improper and should be withdrawn.

Besides not being anticipated by Feichtinger et al., the claims are not rendered obvious by Feichtinger et al. since it does not suggest having inner and outer support locations for two brackets that are located on a common plane that extends at a right angle with respect to the axis of rotation as recited in independent claims 33 and 54. Without such motivation, the claims should be allowed.

CONCLUSION

In view of the arguments above, Applicant respectfully submits that all of the pending claims 12-57 are in condition for allowance and seek an early allowance thereof. If for any reason, the Examiner is unable to allow the application in the next Office Action and believes that an interview would be helpful to resolve any remaining issues, he is respectfully requested to contact the undersigned attorneys at (312) 321-4200.

Respectfully submitted,



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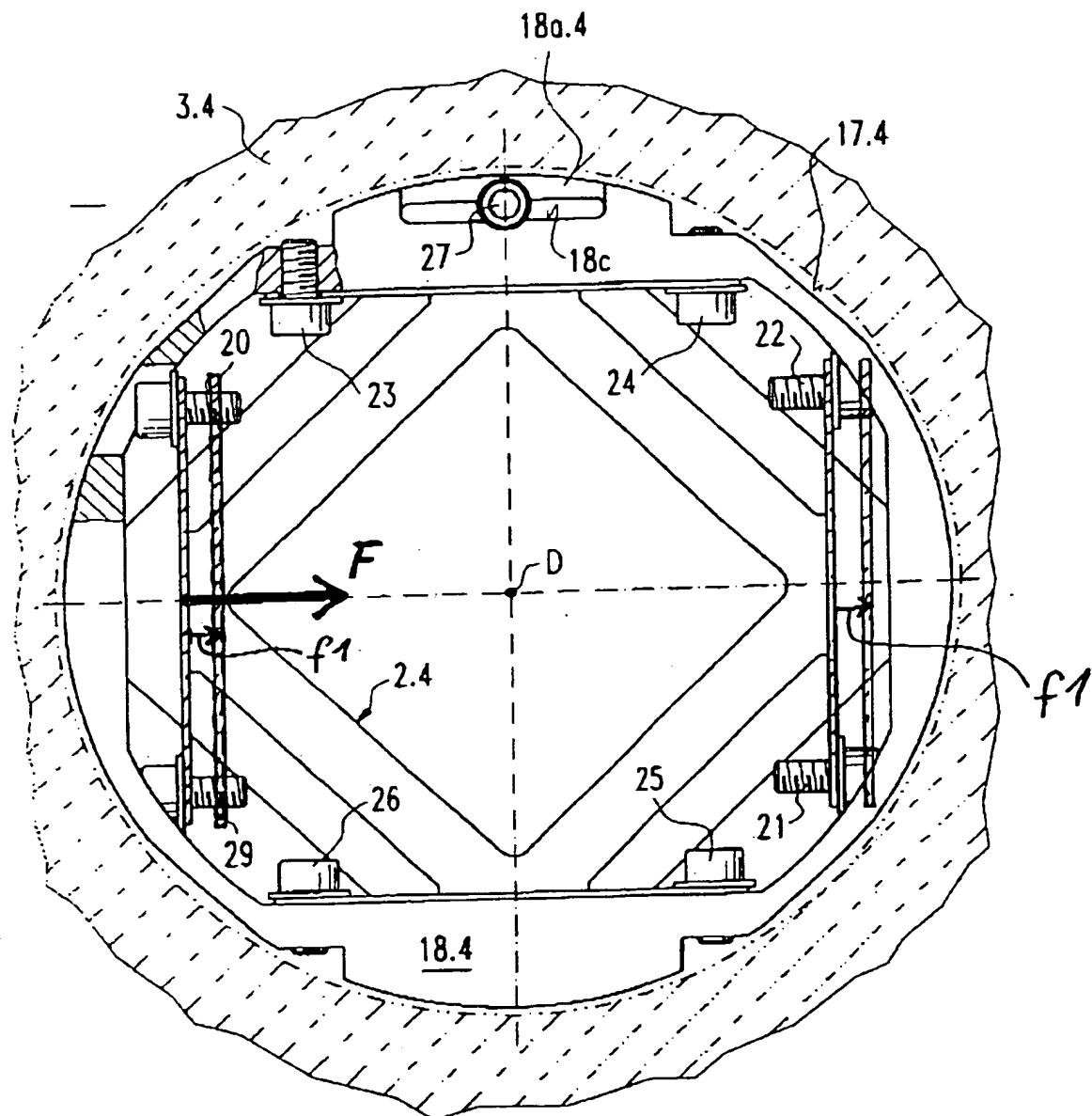


Fig.4

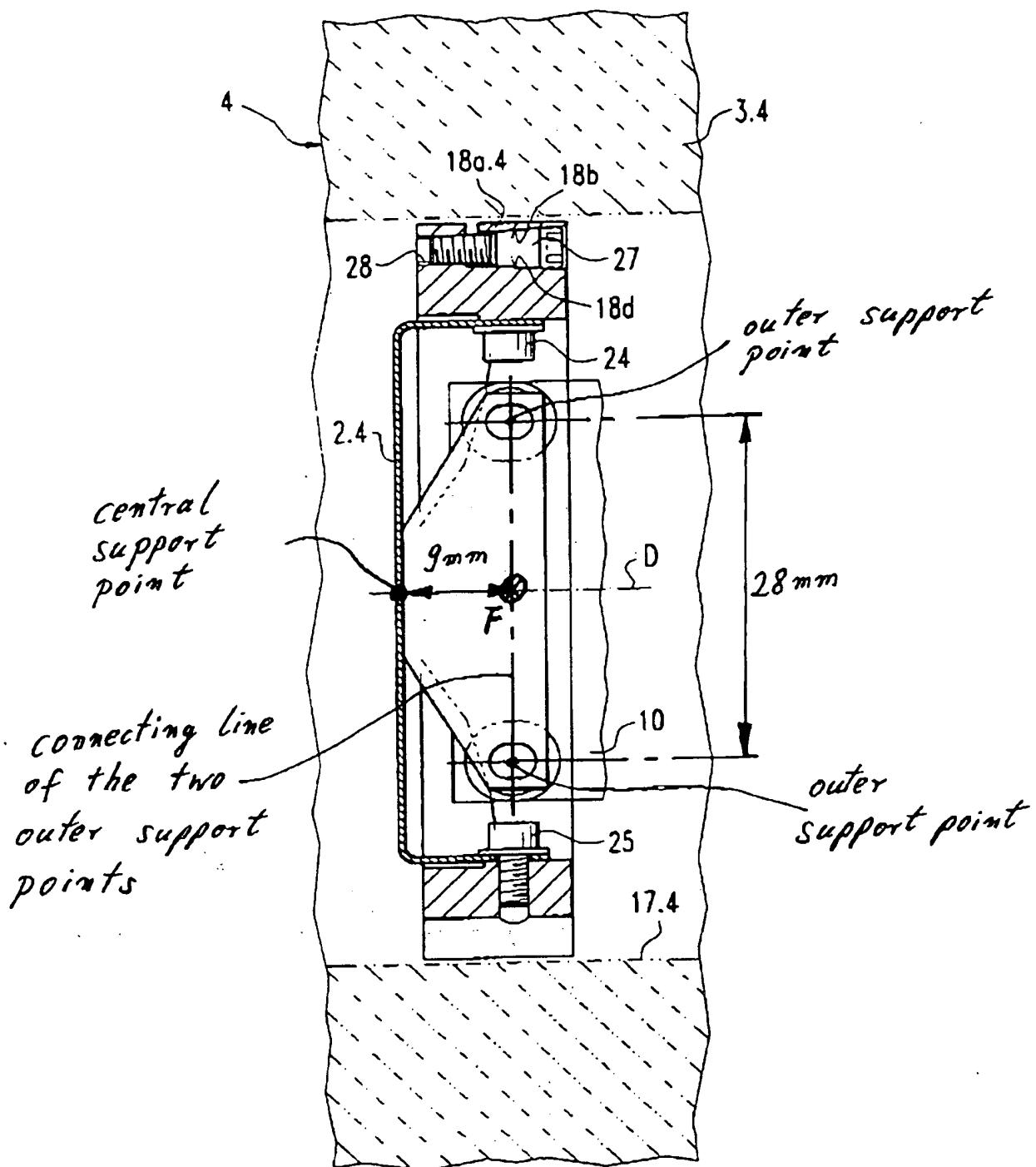


Fig.5

Feinmechanische Bauelemente

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3., stark bearbeitete Auflage

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Zusammenfassend gilt, daß man nur die höchstzulässige Belastung nach Tafel 3.2 benutzt, wenn konstruktive Notwendigkeiten (Raum und Masse) dazu zwingen. Ratsamer ist im Interesse einer höheren Sicherheit, die Beanspruchung niedriger zu wählen.

Je nach der Hauptbelastung des Werkstoffes unterscheidet man Biege- oder Drehfedern und bei diesen wiederum je nach der Halbzeugform solche mit rechteckigem oder mit rundem Querschnitt.

3.2.1.2. Biegefeder

Sie werden in erster Linie auf Biegung beansprucht und in gestreckter Form als Blatt- oder Stabfedern, in gekrümmter Form z. B. als Kontaktfedern und in gewundener Form als Spiralfedern oder als Drehfedern verwendet.

Gestreckte Biegefeder mit rechteckigem Querschnitt (Blattfeder) [49] [53] [58]

Konstruktiv wählt man meist eine einfache Einspannung an einem Ende und die Belastung am freien Ende entsprechend Bild 3.2. Je nachdem, ob die Federbreite von der Eingangsstelle beginnend über die ganze Federlänge gleichbleibend ist oder sich gegen den Wert Null oder einen endlichen Wert verjüngt, spricht man von Rechteckfedern (Bild 3.2), Dreieckfedern (Bild 3.3) oder Trapezfedern (Bild 3.4).

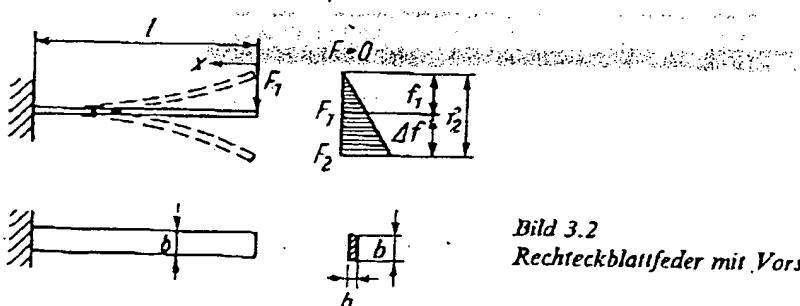


Bild 3.2
Rechteckblattfeder mit Vorspannung F_1

Die Blattfeder kann einlagig oder mit mehreren Lagen ausgeführt werden, wie sie häufig für Kontaktwerke benutzt wird. Kontaktfedersätze für Relais sind standardisiert in [74] bis [76]. Die Verwendung von Blattfedern als Lagerelemente ist aus Abschn. 3.3.1.10. ersichtlich.

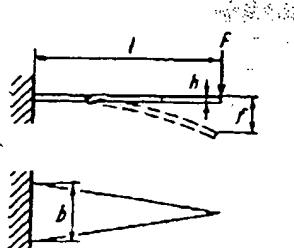


Bild 3.3
Dreieckblattfeder

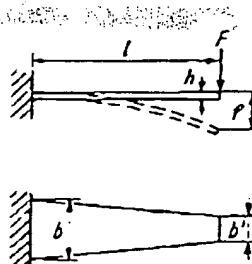


Bild 3.4
Trapezblattfeder

Berechnung

Allgemein gelten für rechteckigen Querschnitt die Beziehungen

Biegemoment

$$M_b = F l_2 \leq W_b \sigma_{b \text{ zul}}; \quad (3)$$

Widerstandsmoment

$$W_b = b h^2 / 6; \quad (4)$$

Flächenträgheitsmoment

$$J = b h^3 / 12; \quad (5)$$

Rechteckfedern (Bild 3.2)

Federkraft

$$F \leq (bh^2/6l) \sigma_{b,zul}; \quad (6)$$

Federweg

$$f = Fl^3/3EJ = 4Fl^3/Ebh^3 \leq (l^2/3Eh) \sigma_{b,zul}; \quad (7)$$

Gesamthub

$$f_2 = f_1 + \Delta f \quad \text{bzw.} \quad f_2 = \Delta f F_2/(F_2 - F_1); \quad (8)$$

Federkonstante

$$c = F/f = bh^3E/4l^3; \quad (9)$$

Federungsarbeit

$$W = Ff/2 = (bhl/18E) \sigma_{b,zul}^2. \quad (10)$$

Für die Biegebeanspruchung gilt

$$\sigma_b = M_x/W_b = Fx/W_b. \quad (11)$$

W_b ist über die ganze Länge konstant, die größte Beanspruchung des Federwerkstoffs tritt also an der Einspannstelle auf ($x = x_{\max} = l$). Das Bestreben, den Werkstoff über die ganze Länge der Feder gleich hoch zu beanspruchen, führt auf die Dreieckfedern mit minimalem Werkstoffaufwand.

$$\sigma_b = 6Fx/h^2b_x \quad (12)$$

bleibt konstant, wenn $b_x \sim x$ wird.**Dreieckfedern (Bild 3.3)**

Federkraft [Gl.(6), s. oben]

Federweg

$$f = Fl^3/2EJ = 6l^3F/Ebh^3 \leq (l^2/Eh) \sigma_{b,zul}. \quad (13)$$

Konstruktiv nachteilig ist das spitze freie Ende der Feder, das keinen Platz für Nocken oder Federaugen bietet. Um diesen Nachteil zu beseitigen, wählt man eine endliche, aber so klein wie mögliche Federbreite am freien Federende und gelangt so zur Trapezfeder.

Hinsichtlich der Werkstoffausnutzung steht diese zwischen den Rechteck- und den Dreieckfedern. Wegen der kleinen Masse der Trapez- und Dreieckfedern gegenüber den Rechteckfedern bei gleichen Federeigenschaften werden diese besonders bei Schnellschwingern angewendet [36].

Trapezfedern (Bild 3.4) [36] [44] [59]

Federkraft [Gl.(6), s. oben]

Federweg

$$f = \psi (Fl^3/3EJ) = 4\psi (Fl^3/Ebh^3) \leq \frac{3}{2} \psi (l^2/Eh) \sigma_{b,zul}. \quad (14)$$

Hierbei ist J auf die Einspannstelle mit Breite b bezogen. Werte für ψ s. Tafel 3.3.

In einigen Fällen stört die bei der Durchbiegung der Feder auftretende Neigung des freien Federendes (Abkippen von Kontaktflächen). Durch parallele Anordnung zweier Blattfedern in der

Tafel 3.3. Trapezfaktor ψ

b'/b	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
ψ	1,500	1,390	1,315	1,250	1,202	1,160	1,121	1,083	1,054	1,025	1,000

Richtung der Durchbiegung wird, wie im Bild 3.5 gezeigt, erreicht, daß sich das freie Ende parallel zur Einspannstelle bewegt.

Fall 1

Parallelfedern

Belastung je Feder mit Kraft F

Federkraft

$$F \leq \frac{1}{3} (bh^2/l) \sigma_{b,zul}; \quad (15)$$

Federweg

$$f = Fl^3/12EJ = Fl^3/Ebh^3 \leq \frac{1}{3} (l^2/Eh) \sigma_{b,zul}. \quad (16)$$

Auch beiderseitig aufgelegte oder eingespannte Federn finden Verwendung.

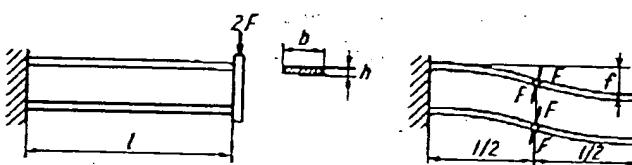


Bild 3.5
Parallelfederanordnung,
unbelastet und belastet

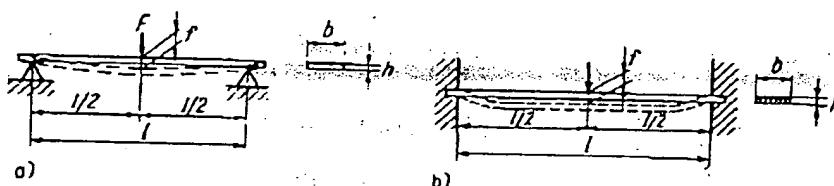


Bild 3.6. Rechteckfeder

a) beiderseitig frei aufgelegt; b) beiderseitig eingespannt

Bei beiderseitig frei aufliegender, in der Mitte belasteter Rechteckblattfeder (Bild 3.6a) gilt (Aufteilung in zwei Halbfedern)

$$F \leq \frac{1}{3} (bh^2/l) \sigma_{b,zul}; \quad (17)$$

$$f = Fl^3/48EJ = Fl^3/4Eb h^3 \leq (l^2/6Eh) \sigma_{b,zul}; \quad (18)$$

bei beiderseitig fest eingespannten Enden (Bild 3.6b)

$$F \leq \frac{1}{3} (bh^2/l) \sigma_{b,zul}; \quad (19)$$

$$f = Fl^3/192EJ = Fl^3/16Eb h^3 \leq (l^2/12Eh) \sigma_{b,zul}. \quad (20)$$

Membranfedern [5]

Als Membran bezeichnet man eine quer zu ihrer Ebene elastisch verformbare Platte. Die geometrische Form ist in den meisten Fällen ein Kreis. In der Literatur der Festigkeitslehre ist sie unter dem Begriff Kreisplatte zu finden.

Membranen werden im Gerätebau als Kraft- bzw. Druckmeß- oder als Führungselemente verwendet.

Für den Belastungsfall mit einer zentralen Einzelkraft F ergeben sich entsprechend Bild 3.7 folgende Beziehungen:

Fall a: eingespannter Rand

$$f = \frac{3Fr^2(1-\nu^2)}{4\pi Eh^3} \quad (21a)$$

Für Metalle mit $\nu = 0,3$ wird $f = 0,22Fr^2/Eh^3$.